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Is pitch system free play and friction important for rotor design?

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Abstract

The loads and the performance of a wind turbine depend on multiple characteristics of the design. Traditional aero-elastic simulation tools include detailed structural and aerodynamic models of the turbines main components: tower, drive train and blades. Crude assumptions regarding smaller components and actuators are often made to limit modeling complexity, and because the influence of these on the global behavior of the turbine is assumed to be small. However, as the turbine design complexity is increased, i.e. with more radical blade design, the influence of component characteristics might become significant. In this study in the influence of pitch gear characteristics on turbine loads and performance is investigated.

Keywords: Pitch system, modeling, loads, performance

1 Introduction

With increasingly larger, flexible and advanced rotor designs including pre-bend, sweep etc., the influence of subcomponent characteristics on the turbine loads, stability and performance is becoming increasingly important. Therefore, there is a gaining interest in aero-elastic simulations that couples subcomponent models to aero-elastic computations.

An example is integrated design of blades and pitch systems. Often the pitch system is modeled as being infinitively torsional stiff. However, in reality, the pitch system has a finite torsional stiffness and might even encompass free-play/backlash. Hence, the effective pitch of the blade is depending on the torsional moment that is applied to the pitch system by the blade. Small variations of the effective pitch of the blade might have significant effect on loads and performance.

In this study, the use of integrated models is exemplified by studying the effect of the free-play/backlash in a pitch gear and the blade bearing friction on loads and performance. The study is based on a coupled model of a pitch system and an aero-elastic turbine model.

In the study, results are presented from two types of electric pitch systems (planetary and cycloidal gear). The systems are based on data supplied by pitch gear manufacturer Nabtesco. The performance of the systems is studied with two types of control applied: Classic collective pitch control and cyclic pitch control.

All simulations are performed in HAWC2 [1] and with a model of the NREL 5MW reference turbine [2] and a generic model of a 1.5MW.

2 Pitch Gear Modeling

An electric pitching system of a horizontal axis wind turbine typically consists of a ring gear mounted at the blade root and a pinion gear attached to an actuator of some sort through a reduction gear. The actuator-reduction gear-pinion assembly is mounted on the hub. When the pinion is rotated by the actuator, the blade pitches. In Figure 1, a schematic of the pitch gear system is shown. In the model shown in Figure 1, the rotational degrees of freedom are the internal rotations of the gears around the steady state pitch value. Thus, the actual pitch of the blade is:

$$\boldsymbol{\theta_p} = \boldsymbol{\theta_p^0} + \Delta\boldsymbol{\theta_p}, \tag{1}$$

where $\Delta\theta_p$ is the pitch increment due to the free play and flexibility of the pitch gear system. It is assumed that the teeth of the ring gear and pinion are infinitively stiff, therefore $\Delta\theta_p$will only depend on the torsional stiffness and damping of the reduction gear k_r and c_r, respectively, that depends on the free-play of the reduction gear τ_r and the free-play of the ring gear-pinion teeth interface τ_g. The effective stiffness and damping of the reduction gear is defined as:

$$\boldsymbol{k}(\Delta\boldsymbol{\theta}_r) = \begin{cases} \boldsymbol{0} & |\Delta\boldsymbol{\theta_r}| \leq \boldsymbol{\tau} \\ \boldsymbol{k_r}\frac{|\Delta\boldsymbol{\theta_r}|-\boldsymbol{\tau}}{|\Delta\boldsymbol{\theta_r}|} & |\Delta\boldsymbol{\theta_r}| > \boldsymbol{\tau} \end{cases} \tag{2}$$

$$\mathbf{c}(\Delta\boldsymbol{\theta}_r) = \begin{cases} \boldsymbol{0} & |\Delta\boldsymbol{\theta_r}| \leq \boldsymbol{\tau} \\ \boldsymbol{c_r} & |\Delta\boldsymbol{\theta_r}| > \boldsymbol{\tau} \end{cases}, \tag{3}$$

where τ is the combined free-play in the reduction gear and between the teeth of the pinion and the ring gear.

Using Newton's 2nd law, the equation of motion of the ring gear is:

$$\boldsymbol{I_g}\ddot{\boldsymbol{\theta}}_{\boldsymbol{p}} - \frac{\boldsymbol{R_g}}{\boldsymbol{R_r}}\boldsymbol{M_r} = \boldsymbol{M}, \tag{4}$$

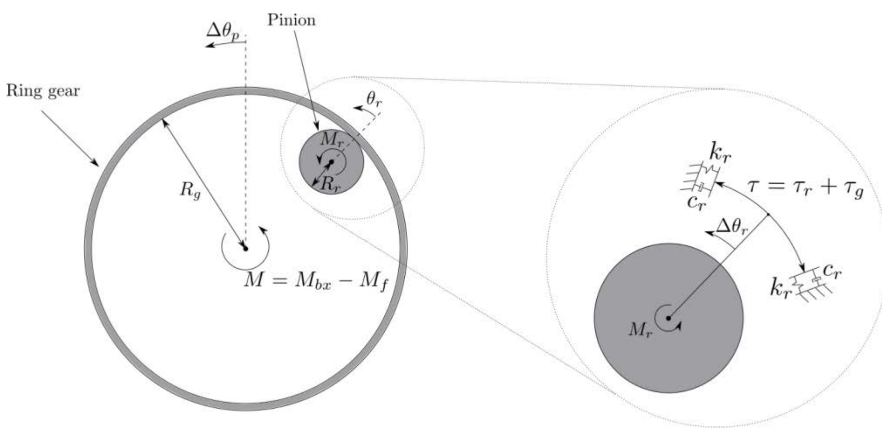


Figure 1: Pitch system model

where I_g is the inertia of the ring around the center axis, R_g is radius of the ring gear, R_r is the radius of the pinion, M_r is the reaction moment of the reduction gear due to a pinion rotation $\Delta\theta_r$, and M is external moments acting on the ring gear.

The reaction moment is defined as:

$$\boldsymbol{M_r} = -\boldsymbol{I_r}\Delta\ddot{\boldsymbol{\theta}}_{\boldsymbol{r}} - \boldsymbol{c}(\Delta\boldsymbol{\theta}_r)\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{r}} - \boldsymbol{k}(\Delta\boldsymbol{\theta}_r)\Delta\boldsymbol{\theta_r}, \tag{5}$$

where I_r is the inertia of the reduction gear. The external moments are defined as:

$$\boldsymbol{M} = \boldsymbol{M_{bz}} - \boldsymbol{M_f}, \tag{6}$$

where $\boldsymbol{M_{bz}}$ is the pitching moment caused by the aerodynamic forces, gravitational loads etc. being transferred to the bearing and $\boldsymbol{M_f}$ is the pitch bearing friction moment.

The motion of the pinion is related to the motion of the ring gear through the following equation:

$$\Delta\boldsymbol{\theta_r} = \frac{\boldsymbol{R_g}}{\boldsymbol{R_r}}\Delta\boldsymbol{\theta_g} \tag{7}$$

Substituting Equations (5), (6), and (7) into (4) yields the following equation of motion of the ring gear:

$$\left(\boldsymbol{I_g} + \frac{\boldsymbol{R_g^2}}{\boldsymbol{R_r^2}}\boldsymbol{I_r}\right)\Delta\ddot{\boldsymbol{\theta}}_{\boldsymbol{p}} + \frac{\boldsymbol{R_g^2}}{\boldsymbol{R_r^2}}\boldsymbol{c}(\Delta\boldsymbol{\theta}_r)\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}} + \frac{\boldsymbol{R_g^2}}{\boldsymbol{R_r^2}}\boldsymbol{k}(\Delta\boldsymbol{\theta}_r)\Delta\boldsymbol{\theta_p} = \boldsymbol{M_{bz}} + \boldsymbol{M_f} \tag{8}$$

or on standard form:

$$\widetilde{\boldsymbol{M}}\Delta\ddot{\boldsymbol{\theta}}_{\boldsymbol{p}} + \tilde{\boldsymbol{C}}\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}} + \widetilde{\boldsymbol{K}}\Delta\boldsymbol{\theta}_{\boldsymbol{p}} = \boldsymbol{M}_{\boldsymbol{bz}}, \tag{9}$$

where

$$\widetilde{M} = \left(I_g + \frac{R_g^2}{R_r^2} I_r\right)$$

$$\widetilde{\boldsymbol{C}} = \begin{cases} \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}} & |\Delta\boldsymbol{\theta}_{\boldsymbol{p}}| \leq \tilde{\boldsymbol{\tau}} \\ \frac{\boldsymbol{R}_{\boldsymbol{g}}^2}{\boldsymbol{R}_{\boldsymbol{r}}^2}\boldsymbol{c} + \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}} & |\Delta\boldsymbol{\theta}_{\boldsymbol{p}}| > \tilde{\boldsymbol{\tau}} \end{cases} \tag{10}$$

$$\widetilde{K} = \begin{cases} 0 & |\Delta\theta_p| \leq \tilde{\tau} \\ \frac{R_g^2}{R_r^2} k & |\Delta\theta_p| > \tilde{\tau} \end{cases},$$

and $\tilde{\tau}$ is the free-play in terms of $\Delta\theta_p$ defined as:

$$\tilde{\boldsymbol{\tau}} = \frac{\boldsymbol{R}_{\boldsymbol{g}}}{\boldsymbol{R}_{\boldsymbol{r}}}\boldsymbol{\tau} \tag{11}$$

The pitch bearing friction is modeled using the Dahl friction model [3]:

$$\frac{\boldsymbol{dM}_{\boldsymbol{f}}}{\boldsymbol{d}\Delta\boldsymbol{\theta}_{\boldsymbol{p}}} = \boldsymbol{\sigma}\left|\boldsymbol{1} - \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\boldsymbol{M}_{\boldsymbol{c}}}\boldsymbol{sign}\left(\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}\right)\right|^{\boldsymbol{n}}\boldsymbol{sign}\left(\boldsymbol{1} - \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\boldsymbol{M}_{\boldsymbol{c}}}\boldsymbol{sign}\left(\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}\right)\right) \tag{12}$$

where $\boldsymbol{\sigma}$ is a stiffness parameter at equilibrium point $\boldsymbol{M}_{\boldsymbol{f}}$ = 0 which in this study is set to $\boldsymbol{\sigma} = \boldsymbol{10^8}$. In the original model $\boldsymbol{M}_{\boldsymbol{c}}$ is the Coulomb friction force. In the present model it is assumed that $\boldsymbol{M}_{\boldsymbol{c}}$ is the friction moment applied on the bearing rather than a force. The friction moment in the pitch bearings of wind turbine is not well documented. In this study the Coulomb friction is assumed to be defined as:

$$\boldsymbol{M}_{\boldsymbol{c}} = \boldsymbol{0.013}\sqrt{\boldsymbol{M}_{\boldsymbol{x}}^2 + \boldsymbol{M}_{\boldsymbol{y}}^2} + \boldsymbol{0.061}\sqrt{\boldsymbol{F}_{\boldsymbol{x}}^2 + \boldsymbol{F}_{\boldsymbol{y}}^2} + \boldsymbol{0.016}|\boldsymbol{F}_{\boldsymbol{z}}| \tag{13}$$

where $\boldsymbol{M}_{\boldsymbol{x}}$ is the flapwise blade root bending moment, $\boldsymbol{M}_{\boldsymbol{y}}$is the edgewise blade root bending moment, F_x is the edgewise blade root force, F_y is the flapwise blade root force, and F_z is the axial force (along the pitch axis). Equation (13) is based on experience. *n* is a material dependent tuning parameter which $0 \leq n \leq 1$ for brittle materials and $n \geq 1$ for more ductile materials. Because of no detailed knowledge of the material parameters *n* is set equal to 1 and Equation (12) is reduced to

$$\frac{\boldsymbol{dM}_{\boldsymbol{f}}}{\boldsymbol{d}\Delta\boldsymbol{\theta}_{\boldsymbol{p}}} = \boldsymbol{\sigma}\left(\boldsymbol{1} - \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\boldsymbol{M}_{\boldsymbol{c}}}\boldsymbol{sign}\left(\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}\right)\right) \tag{14}$$

By dividing on both sides with dt Equation (14) can be rewritten as:

$$\frac{\boldsymbol{dM}_{\boldsymbol{f}}}{\boldsymbol{dt}} = \Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}\boldsymbol{\sigma} - \frac{\boldsymbol{M}_{\boldsymbol{f}}}{\boldsymbol{M}_{\boldsymbol{c}}}|\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}| \tag{15}$$

This model is sufficient to introduce hysteresis in the system. Applying the friction model in the pitch gear model introduces an additional state to the model. On matrix form the derived pitch gear model is defined as:

$$\begin{bmatrix} \widetilde{\boldsymbol{M}} & \boldsymbol{0} \\ \boldsymbol{0} & \boldsymbol{0} \end{bmatrix}\begin{bmatrix} \Delta\ddot{\boldsymbol{\theta}}_{\boldsymbol{p}} \\ \ddot{\boldsymbol{M}}_{\boldsymbol{f}} \end{bmatrix} + \begin{bmatrix} \tilde{\boldsymbol{C}} & \boldsymbol{0} \\ -\boldsymbol{\sigma} & \boldsymbol{1} \end{bmatrix}\begin{bmatrix} \Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}} \\ \dot{\boldsymbol{M}}_{\boldsymbol{f}} \end{bmatrix} + \begin{bmatrix} \widetilde{\boldsymbol{K}} & \boldsymbol{0} \\ \boldsymbol{0} & \frac{\boldsymbol{1}}{\boldsymbol{M}_{\boldsymbol{c}}}\boldsymbol{\sigma}|\Delta\dot{\boldsymbol{\theta}}_{\boldsymbol{p}}| \end{bmatrix}\begin{bmatrix} \Delta\boldsymbol{\theta}_{\boldsymbol{p}} \\ \boldsymbol{M}_{\boldsymbol{f}} \end{bmatrix} = \begin{bmatrix} \boldsymbol{M}_{\boldsymbol{bx}} \\ \boldsymbol{0} \end{bmatrix} \tag{16}$$

2.1 Implementation

In HAWC2 the pitch gear model is implemented as an external system connecting the hubs and the blades of the modeled turbine, see Figure 2. The structural part of HAWC2 is based on a multibody formulation composed of beam elements with 6 degrees of freedom at each node (3 translation and 3 rotations). In the implementation of the pitch gear, it is assumed that only the torsional degree of freedom $\Delta\boldsymbol{\theta}_{\boldsymbol{p}}$ is active. The remaining degrees of freedom are modeled as being stiff.

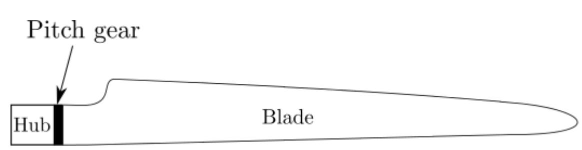


Figure 2: Position of external pitch gear model.

2.2 Model Validation

To test the implemented pitch gear model, simulations are performed without wind. In the simulations, the turbines are simulated in a parked configuration with one blade pointing upwards. A slowly varying torsional moment is applied on the blade side of the gear of the upward pointing blade. The resulting $\Delta\boldsymbol{\theta}_{\boldsymbol{p}}$ is extracted and plotted as a function of the torsional moment on the hub side of the gear. Three gears are tested. The parameters of the tested gears are given in Table 1. Using the tabulated values, the torsional stiffness of the pitch gear can be defined. The inertias of the pitch gear system are neglected, because these are assumed to be small compared to the inertia of the blade. The damping is modeled as being stiffness proportional, $c_r = 0.5k_r$. Figure 3 shows the results for the 5 MW and 1.5 MW turbine, respectively. In both plots, it is seen that there is an overshoot in torsional moment when the bearing reaches the free-play boundaries. Furthermore, it is seen that the kinematic Coulomb friction is larger for the 5 MW turbine than for the 1.5 MW turbine. The difference in Coulomb friction is due to the difference in blade mass and mass distribution of the two turbines.

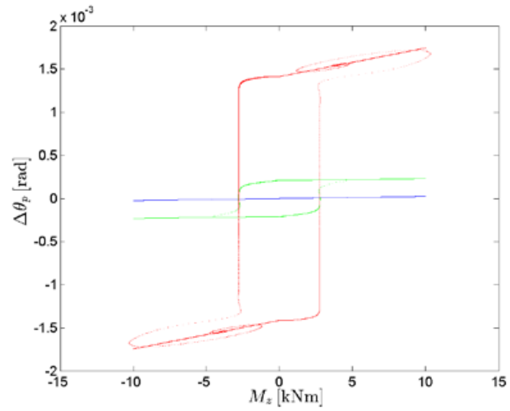
Table 1: Pitch gear parameters, courtesy of Nabtesco

	Free play [deg]	Torsional stiffness [Nm/deg]	Gear ratio
No slip gear	0	86,400	15:139
Cycloidal gear	0.22	86,400	15:139
Planetary gear	1.5	6,120	15:139

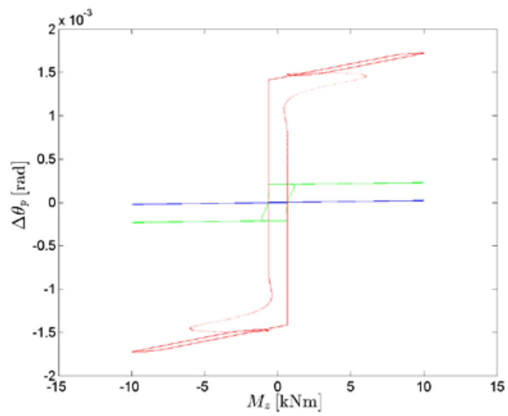
3 Results

Results are presented for two different studies. First, results are presented for simulations of both turbines with the three different pitch gears (see Table 1) in IEC Class C turbulence [4] at wind speeds from 4-24 m/s and power law wind shear with an exponent of 0.2. Then, results are presented for simulations performed on the NREL 5 MW reference turbine with individual pitch applied for load alleviation.

At each wind speed, a 10 minute simulation is performed. In the following, selected results are presented. Figure 4 shows the internal angular position of the gear, $\Delta\theta_p$, for both turbines. For both turbines, the angular displacements are largest for the planetary gear. This also expected because of the less stiffness and larger free-play of this gear. However, it is



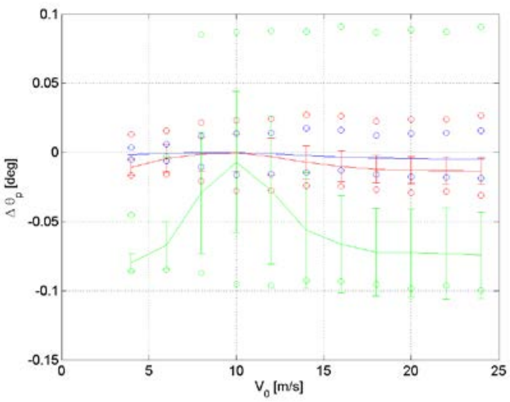
(a)



(b)

Figure 3: Angular displacement of the pitch gear model as a function of torsional moment at the hub when a varying pitching moment is applied to the 5MW reference turbine (a) and the 1.5MW turbine (b). Blue: No slip gear, green: Nabtesco gear, red: Planetary gear.

evident that the gear is loaded differently for the two turbines. For the 5MW turbine the gear position is alternating between being within the free-play and at one of the limits of the free-play. For the 1.5MW turbine the gear is constantly kept at on side of the free-play. The different movement of the gears depending on turbine is due to differences in the aerodynamics of the rotors. The effect of the different gears on the blade root loads are shown in Figure 5 that shows the blade root moments of the two turbines. It is clearly seen that the resulting loads are almost identical. This might seem surprising since the planetary gear is significantly more flexible and has larger free-play that the cycloidal gear. However, as seen from Figure 4 the magnitudes of the additional pitch cause by the gear characteristics, is very small.



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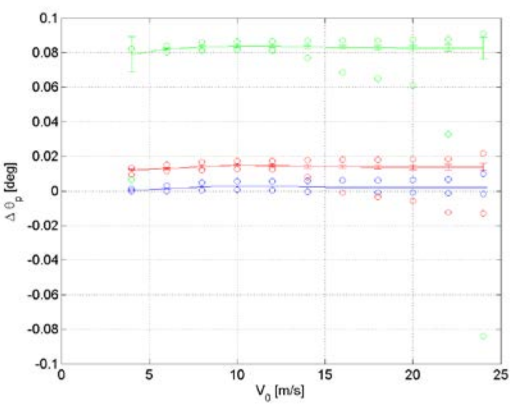
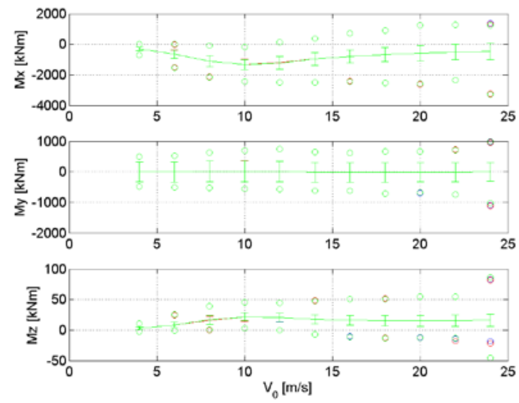
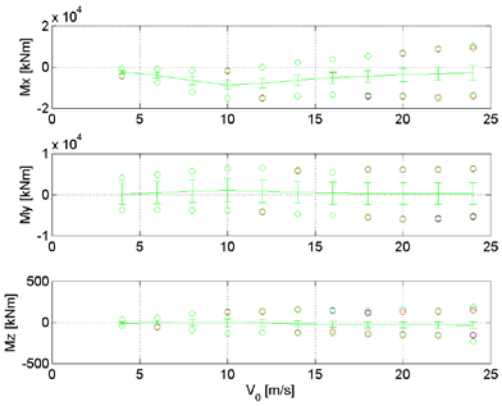


(b)

Figure 5: $\Delta\theta_p$ of the NREL 5 MW reference turbine (a) and the 1.5 MW turbine (b) with the three different pitch gears. Blue: No free-play gear, red: Nabtesco gear, green: planetary gear. The line shows the mean value, the whisker shows the standard deviation, and the bullets represent the min and max values.



(a)

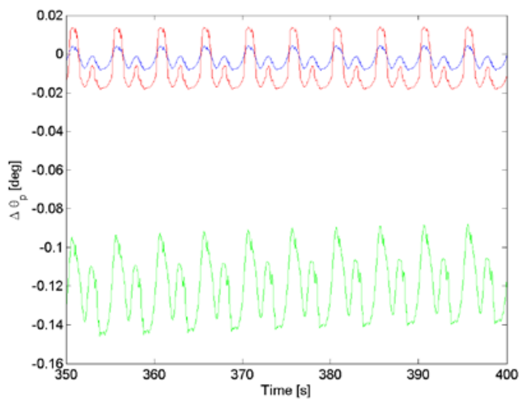


(b)

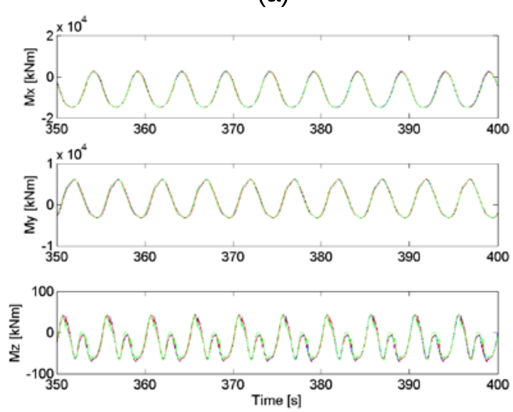
Figure 4: Blade root moments of the NREL 5 MW reference turbine (a) and the 1.5 MW turbine (b) with the three different pitch gears. Blue: No free-play gear, red: Nabtesco gear, green: planetary gear. The line shows the mean value, the whisker shows the standard deviation, and the bullets represent the min and max values.

3.1 Individual Pitch Control

Some modern turbines are operated with individually controlled blades to decrease the load variations on the blades and tower. A study is performed to investigate the effect of the three pitch gears on the performance and loads of an individual pitch controlled turbine. Simulations are performed with the 5MW turbine in deterministic inflow with power law wind shear with an exponent of 0.2 and a mean wind speed of 20 m/s. The applied individual pitch controller is described in [5]. In Figure 6, the results are presented in the form of samples of the time domain simulation results. From the figures, it is seen that even though the mean $\Delta\theta_p$ varies from gear to gear, the effect on the loads is very small.



(a)



(b)

Figure 6: $\Delta\boldsymbol{\theta_p}$ (a) and m**oments (b) of the NREL 5 MW reference turbine with individual pitch at 20 m/s with the three different pitch gears. Blue: No free-play gear, red: Nabtesco gear, green: planetary gear. The line shows the mean value, the whisker shows the standard deviation, and the bullets represent the min and max values.**

4 Suggestions for Further Work

The results presented in this paper suggest that the stiffness characteristics of the pitch gear are not influencing the performance of the turbine. One possible explanation to this could be that the gears are over dimensioned. Thus, a study could be made to assess the lowest allowable gear stiffness using the full aeroelastic simulation coupled with the gear model. By estimating the lower limit, from the results of full aeroelastic simulations coupled with the gear model, the design requirement for the gear can be relaxed, and the cost can potentially be deceased.

Both the turbines that are investigated in the study have a rather conservative blade design, without much bend-twist coupling. Bend-twist coupling can e.g. be achieved through a swept blade design. A swept blade will induce larger torsional moments at the blade root which might increase the importance of the pitch gear characteristics for the turbine performance. This could be investigated by studying turbines with swept blades.

Finally, the pitch motor dynamics are not included in the current model. This could be included to yield an even more accurate model.

5 Conclusions

This paper raises the question "Is pitch system free play and friction important for rotor design?". This was investigated by implementing a nonlinear stiffness model of two types of electrical pitch gear systems in the aeroelastic simulation tool HAWC2. The results showed that the performance of the turbine is only marginally affected by changes of the pitch gear. Thus, the short answer to question to the question is no, the pitch gear characteristics does not appear to be important for the rotor design. However, the presented results are for turbines with a conservative blade design. It is plausible that blades with large bend-twist coupling can yield a greater importance of the pitch gear characteristics. In this paper a framework for testing the influence of subcomponent characteristics where developed. The use of the framework was exemplified by studying a pitch gear. However, the framework could be used for including detailed models of other types of subcomponents.

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